# RESEARCH MEMORANDUM

EXPERIMENTAL INVESTIGATION OF EXTERNAL WATER-SPRAY COOLING IN A TURBOJET ENGINE UTILIZING SEVERAL INJECTION CONFIGURATIONS INCLUDING ORIFICES IN THE ROTOR-BLADE BASES

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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## SUMMARY

A representative centrifugal-flow turbojet engine was modified to permit water injection from orifices located in the bases of turbine rotor blades and from stationary orifices located in the stator diaphragm. An experimental investigation was conducted with this engine to determine the cooling effectiveness of water injection from rotating injection orifices, alone, or in combination with stationary injection orifices.

The engine was operated with spray cooling over a range of speeds up to rated (11,500 rpm) and turbine-inlet gas temperatures up to 1700° F. The most effective cooling was obtained with a configuration comprising a combination of rotating and stationary injection orifices. This configuration consisted of five 0.031-inch-diameter injection orifices in the rotor-blade base spaced approximately equally along the suction surface and two 0.200-inch-diameter orifices located in the inner ring of the stator diaphragm near the stator-blade trailing edges. At rated engine speed (11,500 rpm) and 1700° F turbine-inlet gas temperature, virtually flat chordwise temperature distributions were obtained with this configuration at all spanwise stations. All measured blade temperatures were below 500° F, and the tip trailing-edge temperature was 280° F. These results were obtained with a total equivalent coolant-to-gas flow ratio of 0.036 with 0.024 of this total passing through the rotating orifices and 0.012 through the stationary orifices.

Spray-cooling operation showed that location of rotating injection orifices along the suction surface side of the blade was more favorable than location along the pressure surface.

Operation with rotating injection orifices alone resulted in satisfactory cooling of only the midchord and the trailing-edge regions of the blade, with a maximum temperature difference between the leading edge and the midchord of 750° F.

## INTRODUCTION

Injection of water sprays into the gas stream of turbojet engines is being investigated at the NACA Lewis laboratory as a means of externally cooling turbine rotor blades to permit turbine operation at higher levels of gas temperature and blade stress. Impingement of spray particles on the hot blade surfaces results in high evaporative heat-transfer rates and extremely effective cooling of the blade portions adequately covered by spray film.

Several factors must be considered before spray-cooling may be applied to an aircraft turbine-engine installation:

(1) If spray cooling is to be of value, the turbine must be operated at gas temperatures and speeds above those feasible without cooling, so that thrust increases may be realized (ref. 1).

(2) Turbine operation over a range of gas temperatures at a high speed requires a substantial operating margin between the compressor operating point and the compressor surge region.

(3) Turbine operation at high rotational speeds requires that the turbine disk and the compressor withstand the elevated stress levels.

(4) Since the coolant supply is limited and the coolant is not recovered, spray cooling is restricted to short periods, such as during take-off, climb, or combat manuever.

(5) The rotor-blade material must withstand thermal shock and stresses induced by the sudden impingement of water sprays upon heated rotor blade surfaces.

(6) Nonuniform blade spray coverage resulting in failure-inducing chordwise blade-temperature differences cannot be tolerated.

Some of these factors have been investigated in the spray-cooling research program in progress. Results are reported in references 1 to 3. Briefly, a theoretical analysis showed that engine thrust might be increased up to 40 percent above rated thrust if turbine operation at a speed of 110 percent of rated value and a gas temperature of 2000° F were permitted. A representative centrifugal-flow turbojet engine was subsequently modified for spray cooling. Spray cooling at rated engine speed reduced blade temperature greatly, but also induced significant chordwise blade-temperature differences. These differences caused several blade failures (ref. 1). To reduce the blade-temperature differences, various methods of water-spray injection from stationary orifices were systematically investigated. Large chordwise temperature differences at the blade root and the midspan were eliminated at

coolant-to-gas flow ratios of approximately 0.03 by using a configuration of orifices located in the inner ring of the stator diaphragm near the stator-blade trailing edges (ref. 2). Chordwise temperature differences at the blade tip persisted, however. Despite these tip differences, a later-model centrifugal-flow engine was modified for engine test to illustrate the gains in engine performance that might be attained at overspeed and overtemperature with spray cooling. Substantial gains in thrust were obtained, but blades repeatedly failed at the tip in the region of the large chordwise temperature differences (ref. 3).

Since water injection from stationary injection orifices alone did not adequately cool the blade-tip region (ref. 3), a system to inject coolant from orifices located in the turbine-blade bases was designed. The turbine of a centrifugal-flow turbojet engine was modified to permit water injection from orifices located in the bases of four test blades. The most effective stationary injection configuration that could be designed from previous experience was also provided in the test engine. An experimental investigation was initiated to determine the cooling effectiveness of the rotating injection orifices alone, and in combination with stationary injection orifices. The results of this investigation are reported herein. Engine operation with spray cooling was conducted over a range of speeds up to rated speed (11,500 rpm), turbine-inlet gas temperatures up to 1700° F, and equivalent total coolant-to-gas flow ratios up to 0.036 (coolant-flow rate of approximately 18 gal/min). The term equivalent coolant-to-gas flow ratio is employed throughout the text wherever results involving the rotating injection system are discussed. Since only four test blades were cooled by the rotating system, values of coolant flow are much less than would be required for an entire set of blades. The stationary injection system cooled the entire set of blades. In order to evaluate the rotating injection system with respect to the stationary injection system, an equivalent flow equal to that required for cooling the entire set of blades with the rotating injection system was used in computing coolant-to-gas flow ratio. Total gas flow through the turbine was always used in determining this ratio.

## APPARATUS

The engine used was the same model as that described in references 1 and 2, and the test installation was generally similar to that described in these references. The turbine rotor blades were S-816 alloy.

### Engine Modifications

Rotating injection system. - Four blades 90° apart were cooled by water injected from the rotating injection system. A sectional view of a portion of the engine (fig. 1) illustrates the components of the

rotating injection system. The system consists of a stationary supply manifold, a rotating gutter, transfer tubes that lead from the gutter to axial passages in the rotor rim, flow-control orifices in the rotor rim immediately above these axial passages, and injection orifices in the blade bases. Water was supplied to the stationary manifold through stainless-steel tubing inserted into the tail cone through hollow tail-cone supporting struts. A machined V-type gutter was affixed by screws to a thermocouple-junction terminal block attached to the turbine rotor. A cover plate served to prevent moisture from entering the area where the thermocouple junctions were located. Absence of turbine-blade thermocouple instrumentation, as in possible commercial applications, would, of course, simplify the installation. Water was discharged from the stationary manifold into the rotating gutter through four ejection tubes. In accordance with unpublished NACA results and data from reference 4, the tubes were so bent as to provide a $20^{\circ}$ angle with the plane of rotor rotation as well as tangential water entry into the gutter. In this manner, satisfactory transfer of water from stationary to rotating components was achieved.

After collection in the rotating gutter, the water flowed through transfer tubes strap-welded to the rear face of the rotor to the axial passages in the rotor rim. The transfer tubes were rigidly attached to the gutter by soldering, but were left free at the other end to expand within blocks that covered the axial water passages. Complete application would, of course, require axial water passages under each blade and a second gutter on the rotor rim, connected to these passages. The second gutter would be fed from the first through similar transfer tubes. To insure equal distribution through the four transfer tubes used, a uniform restriction was placed in each tube at its junction with the gutter. The maximum water-flow rate for any rotative speed was established by the metering action of orifices 0.015-inch in diameter located in plugs that were force-fitted into the rotor above the rotor axial passages. These orifices were readily fabricated by conventional drilling procedures. They provided the minimum flow area in the coolant system and may therefore be considered as the flow-control devices. Since the plugs extended into the region between the blade and rotor, holes were provided in the blade locking straps to accommodate the plug extensions. It may be noted from figure 1 that a slight radial clearance exists between the blade base and the plug extension. To minimize the possibility of leakage due to misalinement of the flow passages, each blade-base passage was countersunk. Since coolant is only in use during engine operation, centrifugal force on the fluid is adequate to permit it to bridge the slight clearance space, thereby also preventing axial leakage.

Details of the blade construction are shown in figure 2. The bottom serration on each test blade was removed. Five 1/8-inch-diameter passages were burned out by the intermittent arc process in the blade

base and finish-drilled to size. Machined plugs with 0.031-inch orifices drilled to eject water in the desired direction were fitted into the outer ends of these passages and brazed in place. With all orifices, the direction of water ejection was such as to cause the water to strike the blade surface approximately 1/2-inch above the base, as checked in a static test. The two rotating injection configurations investigated are illustrated in figure 2. Configuration 1 consisted of five injection orifices along the blade suction surface, and configuration 2 consisted of five injection orifices, four along the pressure surface and one near the leading edge of the suction surface. A photograph of the turbine rotor assembled in the engine (fig. 3) further illustrates the rotating injection system.

To minimize the possibility of clogging the small flow-control orifices, distilled water was used in the rotating injection system.

Stationary injection system. - The stationary injection configurations were similar to those described in reference 2. The stationary injection orifices were located in the inner ring of the stator diaphragm near the stator-blade trailing edges. Axial location of these orifices with respect to various parts of the engine is shown in figure 1. Orifices with diameters of 0.200, 0.150, 0.078, and 0.052 inch were provided in pairs. A pair consisted of two nozzles of the same diameter. They were located on the inner ring of the stator diaphragm 180$^{\circ}$ apart. Water flowed through these orifices from tubes welded to the stator diaphragm. City water was used in the stationary injection system.

## Instrumentation

The general engine instrumentation is described in detail in reference 3.

Liquid measurements. - Rates of fuel and water flow were measured by calibrated rotameters. Separate rotameters measured the rates of water flow through the stationary and the rotating injection systems.

Rotor-blade instrumentation. - A total of 12 thermocouples was installed on the test blades; 3 on each of the four blades. The 12 locations are shown in figure 4. As in previous investigations (refs. 1 to 3), a slip-ring pickup was used to provide electric connection between the rotating thermocouples and a potentiometer.

# PROCEDURE

## Design Calculations

The major design limits for the rotating injection system were as follows:

(1) Maximum equivalent coolant weight flow at rated-engine-speed conditions was set at 9200 pounds per hour. Flow was to be apportioned to the four blades selected for cooling on the basis of 9200 pounds per hour for the entire set of blades.

(2) The minimum flow area of the system should not be exposed to the gas stream. This specification was made in order to minimize clogging of the flow-control orifices, and thereby reduce the possibility of excessive build-up of a liquid column that might cause axial leakage between the blade base and the turbine disk. The flow-control, or minimum-area, orifices were located well upstream of the injection orifices.

(3) The injection-orifice diameter was set at 0.031 inch because orifices of this size previously exposed to a combustion-gas stream had not clogged during extended periods of engine operation.

Conventional fluid flow equations were used to determine the pressure required to pass the design flow through the flow-control orifices. Nonrigorous assumptions were made to simplify the design calculations. An orifice coefficient of 0.6, recommended in reference 5 for sharp-edged circular orifices with low throat-to-pipe diameter ratios, was employed. Pressure drop across the injection orifice was neglected and pressure downstream of the flow-control orifices was consequently assumed to be equal to the average value of the gas static pressure at the rotor. As a result of these generalizations, the calculated value of pressure required was approximate. The calculated pressure head was, however, many times smaller than the 4-inch length of the liquid transfer tubes (fig. 1). As a result, a safety factor was provided sufficient not only to account for the nonrigorous assumptions employed in the design calculations but to permit considerable latitude in permissible flow rates.

## Engine Operation

The complete range of engine operating conditions reported in this investigation is presented in table I. Each rotating injection configuration was investigated without the stationary injection system over a range of speeds from 5000 to 11,500 rpm (rated speed). At rated speed, each rotating injection configuration was also investigated in

combination with stationary injection configurations. At rated engine speed, a range of inlet-gas temperatures from 1600° to 1700° F was achieved by varying the exhaust-nozzle setting. At speeds below rated, all operation was conducted with full-open exhaust nozzle. A range of total equivalent coolant-to-gas flow ratios from zero to 0.036 was covered. When combined rotating and stationary injection was employed, the total equivalent coolant-to-gas flow ratio was based on the equivalent flow through the rotating system plus the flow through the stationary system. A run was made at 8000 rpm with the most favorable stationary injection configuration of reference 3 by itself, to provide further comparison with rotating injection. During each cooling run, the water spray was turned on simultaneously with the engine start to minimize thermal shock of the blades.

## RESULTS AND DISCUSSION

Blade-temperature distributions obtained with the various injection configurations investigated, spray deposit patterns, and spray-cooling operating characteristics are presented and discussed in the following sections.

### Blade-Temperature Distributions

Blade-temperature data were obtained for each injection configuration and at each coolant flow used. In order to facilitate comparison of the resulting large number of temperature distributions, only the most favorable of each group are presented. To provide comparison with uncooled operation, the blade-temperature distribution obtained without cooling is shown in each case. Temperature distributions obtained at rated speed with the two rotating injection configurations investigated are presented first. The more favorable rotating injection configuration is then compared at rated speed with the most favorable stationary injection configuration, as determined from earlier investigations. Next, a comparison of the temperature distributions obtained at a speed less than rated with the most favorable rotating and stationary injection configurations at the same equivalent coolant-to-gas flow ratio is presented. Finally, the combination of a rotating configuration and a stationary configuration that was best at rated speed is presented. This presentation includes the effect of varying the distribution of coolant flow between the stationary and the rotating configurations of this combination while maintaining a constant total equivalent coolant-to-gas flow ratio at rated speed.

<u>Injection from rotor-blade orifices</u>. - The blade-temperature distributions obtained at rated speed and maximum coolant flow with the two rotating injection configurations are presented in figure 5. The equivalent coolant-to-gas flow ratio was 0.035 and the inlet-gas

temperature was 1576° F. Figure 5 indicates generally similar chordwise temperature patterns at all spanwise stations. These patterns show high leading-edge temperatures and low trailing-edge temperatures. Configuration 1, with injection along the suction surface, is the more favorable of the two configurations, showing approximately 500° F lower temperatures at the heretofore difficult to cool (refs. 1 to 3) tip trailing-edge section. The trailing-edge temperature obtained with configuration 1 was approximately 300° F at all spanwise stations. Despite the satisfactory cooling obtained at the blade midchord and trailing-edge sections, the leading-edge temperatures of approximately 1000° F contributed to chordwise temperature differences as high as 750° F. These are as great as those encountered in previous spray-cooling investigations with stationary injection configurations. Consequently, water injection from orifices in the rotor-blade bases alone does not appear to be a completely satisfactory system.

Comparison of injection from rotor blade and stationary orifices. - The rated-speed blade-temperature distributions obtained with the more favorable rotating and most favorable stationary injection configurations are illustrated in figure 6. Both configurations are compared at an equivalent coolant-to-gas flow ratio of approximately 0.035. The most favorable stationary injection configuration determined in a previous investigation (ref. 3) consisted of two 0.200-inch-diameter and two 0.052-inch-diameter orifices located in the inner ring of the stator diaphragm near the stator-blade trailing edges. The stationary injection configuration showed virtually flat chordwise temperature distributions at the root and midspan sections. At the blade tip however, the trailing-edge temperature was 850° F and the temperature difference between the trailing edge and the blade midchord was approximately 500° F. The rotating injection configuration did not provide a flat temperature distribution at any spanwise station. Leading-edge temperature was about 1000° F at each station, and the maximum temperature difference between the leading edge and midchord was 750° F (at the blade tip). The trailing-edge and midchord regions were adequately cooled, however, at all spanwise stations. This chordwise distribution of temperature at the blade tip is the reverse of that encountered with stationary injection. The comparison indicates that at relatively high blade-tip speeds (1305 ft/sec) neither method of injection is satisfactory when used by itself, because of the large stress-inducing chordwise temperature differences.

A similar comparison between the same rotating and stationary injection configurations at an engine speed of 8000 rpm (908 ft/sec tip speed) and the same coolant-flow ratio of 0.035 is shown in figure 7. A flat chordwise temperature distribution occurred at all spanwise stations with the stationary injection configuration. The rotating injection configuration showed patterns of chordwise blade temperature at the midspan and tip similar to those present at rated speed. Leading-edge temperatures were about 900° F, and temperature differences

between the leading edge and the midchord were about 700° F. Only the blade root section had a fairly flat temperature distribution. Consequently, it appears that for some engine operating conditions spray cooling from stationary injection orifices is satisfactory, whereas injection from orifices in the rotor blade bases is not.

Injection from a combination of rotor-blade and stationary orifices. - In view of the results obtained from separate operation with rotating and with stationary injection configurations, it appeared that operation with a combination of these configurations might eliminate chordwise temperature differences. Figure 8 presents the temperature distributions observed at rated speed with the most favorable combination of these configurations at a total equivalent coolant-to-gas flow ratio of 0.035. The rotating injection portion of this combination consisted of five orifices in the rotor-blade base along the suction surface, previously discussed as rotating configuration 1. The stationary injection portion of this combination consisted of two 0.200-inch-diameter orifices located in the inner ring of the stator diaphragm at the stator-blade trailing edges. An equivalent coolant-to-gas flow ratio of 0.023 was passed through the rotating injection system and a flow ratio of 0.012 was passed through the stationary system. Figure 8(a) presents the temperature distribution obtained during engine operation with an open exhaust nozzle, which provided a turbine-inlet gas temperature of 1600° F. A virtually flat chordwise temperature distribution was observed at all spanwise stations. Maximum leading-edge temperature was 400° F, and maximum trailing-edge temperature was 260° F. At a higher turbine-inlet gas temperature (1700° F), flat chordwise temperature distributions were also observed (fig. 8(b)). In this instance, maximum leading-edge temperature was 430° F, and maximum trailing-edge temperature was 280° F. All measured blade temperatures were below 500° F at a total equivalent coolant-to-gas flow ratio of 0.036, with 0.024 of this total equivalent passing through the rotating orifices and 0.012 through the stationary orifices.

The injection configuration that permitted operation without troublesome chordwise temperature differences at all spanwise stations was not a combination of the most satisfactory stationary and the most satisfactory rotating injection configurations. As indicated previously, the former consisted of two large (0.200-in. diameter) and two relatively small (0.052-in. diameter) orifices, whereas, only the two large orifices comprised the stationary injection portion of the combination rotating and stationary injection configuration discussed previously.

Penetration of a liquid jet into a gas stream is dependent upon momentum of the jet (mass times velocity). For a given supply pressure, velocity through orifices of any diameter is the same but mass flow through a larger orifice is greater. Consequently, the product of the two terms (momentum) is greater for the larger orifice, and penetration

of the issuing jet is greater. Since the rotating injection configurations investigated could not adequately cool the blade leading-edge region at the tip, a large stationary injection orifice was required to provide sufficient penetration into the gas stream to cool the tip leading-edge region. With a given amount of coolant flow available, use of small stationary injection orifices in conjunction with the large diameter orifices merely decreased the mass flow available to the large orifices and decreased the degree of penetration, resulting in inadequate cooling at the blade tip. Since the leading edge at the blade root was satisfactorily cooled with the large orifices, when used in conjunction with rotating injection, the need for small stationary injection orifices was eliminated.

Since flow rates through the rotating and the stationary portions of a combination injection configuration may be varied while maintaining total flow constant, optimum flow distribution should be determined. The results shown in figure 8 represent near optimum flow distribution for this configuration, as illustrated in figure 9. Figure 9 shows the effect of three changes in the amounts of flow to the stationary system and to the rotating system, for a constant value of total equivalent coolant-to-gas flow ratio of 0.035. With equal equivalent flow rates through the rotating system and through the stationary system, temperatures of the leading-edge and trailing-edge regions of the blade were considerably lower than those of the midchord region. When the rotating system equivalent flow rate was twice that of the stationary system, blade chordwise temperature differences at each spanwise station were comparatively small. When the rotating system equivalent flow rate was approximately five times that of the stationary system, the blade leading-edge temperature, particularly at the blade tip, was higher than with either of the other coolant-flow distributions. Although not subject to high centrifugal stresses, the blade tip region cannot tolerate large chordwise temperature differences, as pointed out in reference 3. Since it is desirable that chordwise temperature differences at all spanwise stations be small, from considerations of blade stress reduction, the 2:1 combination of equivalent flow rates may be considered approximately optimum for the injection configuration under consideration, at rated-engine-speed conditions. It is the one for which the data of figure 8 were plotted.

## Spray Deposit Patterns

Spray deposit patterns have been useful in previous investigations (refs. 1 and 2) as a means of providing a visual method of interpreting the blade-temperature distributions observed. Spray deposit patterns consist of mineral deposits from the water, and occur on portions of the blade surface not thoroughly covered by a liquid film. Figure 10 is a photograph of a section of the turbine rotor showing several turbine

blades after rated-speed spray-cooled engine operation with the most favorable injection configuration consisting of a combination of rotor-blade and stationary orifices. One of the four test blades with coolant injection orifices in the base is shown. This blade is readily identified by the noticeable lack of a spray deposit pattern over the trailing-edge region. The adjacent blades, cooled only through the stationary injection orifices in the stator diaphragm, display a clearly defined area of deposition over the trailing-edge region. Lack of such an area of deposition over the trailing-edge region of the test blade indicates that a film of liquid, apparently supplied from the orifices in the blade base, was sufficient to wet, and, conseuqnetly, cool this area. The blade-temperature distributions previously shown verify this indication. The small, vague pattern near the midchord portion of the test blade is considered insignificant, in view of the low blade temperatures observed in this region and the thinness of the deposit as compared with thick areas of deposition on the other blades.

## Spray-Cooling Operating Characteristics

Blade thermal stress. - The temperature-distribution curves presented herein show that temperatures of spray-cooled blades are drastically lower than those of uncooled blades. Previous experimental investigations (refs. 1 to 3) indicated, however, that large chordwise temperature differences with attendant failure-inducing thermal stresses were generally caused by spray-cooling. Unfavorable chordwise temperature differences have been almost completely eliminated by use of a combination of stationary and rotating injection orifices as reported herein. Elimination of these temperature differences was also accompanied by a low level of cooled-blade temperature. For example, the uncooled-blade temperatures were approximately 1000° F higher than the cooled-blade temperatures (fig. 8). Attempts to operate more efficiently by reducing total coolant flow to raise the level of the cooled-blade temperature uniformly were unsuccessul. Since large chordwise temperature differences will severely reduce blade life, some degree of overcooling apparently must be tolerated if spray-cooling is to be employed.

Leakage. - In a rotating injection system, there are several locations at which leakage may occur. In the design described herein, liquid transfer to the rotating gutter was achieved without spillage by so locating the manifold ejection tube as to provide tangential entry of the liquid to the gutter. Overflow of the gutter would occur, of course, if the quantity of liquid supplied to the gutter were greater than the carry-off-capacity of the system. The capacity of the gutter and the system leading to the injection orifices was sufficient that such a condition did not occur. A second area of possible leakage is around the liquid-transfer blocks. These blocks (fig. 1) were originally welded into position; slight leakage developed after prolonged engine operation. A silver-solder overlay prevented further leaks at this

point. A third location of possible major leakage is through the space between the blade base and the turbine disk. There was no evidence of leakage at this station. In normal engine operation, centrifugal force causes the liquid to bridge the slight gap between disk and blade. Misalinement between the disk and the blade passages would cause axial leakage at this point and care should be taken during fabrication to avoid this possibility. If a column of liquid were to build up in the blade-base passage to the locking straps, axial leakage would occur. Such leakage becomes a possibility if the injection orifice becomes clogged. This condition did not occur during the present investigation, and seems unlikely since the major restriction in the system, the flow control orifice, is upstream of this location. The results of the present investigation indicate that use of a rotating injection system need not cause excessive mechanical difficulties or leakage.

## SUMMARY OF RESULTS

The following results were obtained from an investigation of turbine-blade external spray cooling conducted to determine the cooling effectiveness of water injection from rotating injection orifices alone or in combination with stationary injection orifices.

1. For spray cooling using only injection orifices in the rotor-blade bases, it is more effective to locate the orifices along the suction surface side of the blade than along the pressure surface side.

2. Operations with the more favorable rotating injection configuration alone provided satisfactory cooling of the blade midchord and trailing-edge regions, but the leading edge was as much as 750° F hotter than the midchord.

3. Satisfactory cooling of the entire blade could not be achieved by water injection from rotating injection configurations alone.

4. Satisfactory cooling was achieved with an injection configuration consisting of a combination of rotating and stationary injection orifices. This configuration consisted of five 0.031-inch-diameter orifices in the rotor-blade base along the suction surface and two 0.200-inch-diameter orifices located in the inner ring of the stator diapnragm near the stator-blade trailing edges. Virtually flat chordwise temperature distributions were obtained with this configuration at all spanwise stations, and all measured blade temperatures were below 500° F. The total equivalent coolant-to-gas flow ratio was 0.036.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, December 2, 1954

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TABLE I. - RANGE OF OPERATING CONDITIONS

| Injection configuration | | Nominal engine speed, rpm | Turbine-inlet gas temperature, °F | Equivalent coolant-to-gas flow ratio | |
|---|---|---|---|---|---|
| Orifices, size, and location | Designation | | | Rotating injection system | Stationary injection system |
| Rotating-injection operation | | | | | |
| Five 0.031-in. - suction surface of rotor blades | 1-R | 5,000 | 1118 | 0.0251 | 0 |
| | | 8,000 | 1097 | .0249 | |
| | | 10,000 | 1241 | .0252 | |
| | | 11,500 | 1577 | .0251 | |
| | | 5,000 | 1118 | .0340 | |
| | | 8,000 | 1097 | .0340 | |
| | | 10,000 | 1229 | .0336 | |
| | | 11,500 | 1576 | .0344 | |
| Five 0.031-in. - pressure surface of rotor blades | 2-R | 5,000 | 1235 | 0.0259 | 0 |
| | | 8,000 | 1122 | .0252 | |
| | | 5,000 | 1140 | .0343 | |
| | | 8,000 | 1124 | .0345 | |
| | | 10,000 | 1232 | .0336 | |
| | | 11,500 | 1515 | .0348 | |
| Stationary-injection operation | | | | | |
| Two 0.200-in. and two 0.052-in. - inner ring of stator diaphragm near stator-blade trailing edges | 1-S | 8,000 | 1124 | 0 | 0.0346 |
| | | 11,500 | 1660 | 0 | .0346 |
| Two 0.150-in. and two 0.078-in. - inner ring of stator diaphragm near stator-blade trailing edges | 2-S | 11,500 | 1680 | 0 | 0.0347 |
| Two 0.200-in. - inner ring of stator diaphragm near stator-blade trailing edges | 3-S | 11,500 | 1699 | 0 | 0.0287 |
| Combined rotating- and stationary-injection operation | | | | | |
| | 1-R and 2-S | 11,500 | 1600 | 0.0234 | 0.0116 |
| | | 11,500 | 1636 | .0235 | .0175 |
| | 1-R and 3-S | 11,500 | 1612 | 0.0175 | 0.0175 |
| | | | 1608 | .0231 | .0120 |
| | | | 1590 | .0289 | .0058 |
| | | | 1706 | .0238 | .0118 |
| | | | 1645 | .0232 | .0115 |
| | | | 1624 | .0271 | .0116 |
| | | | 1647 | .0271 | .0174 |
| | 2-R and 3-S | 11,500 | 1547 | 0.0231 | 0.0117 |
| | | | 1550 | .0346 | .0114 |
| | | | 1565 | .0348 | .0173 |
| Uncooled operation | | | | | |
| | | 8,000 | 1140 | ------ | ------ |
| | | 11,500 | 1597 | ------ | ------ |
| | | 11,500 | 1670 | ------ | ------ |

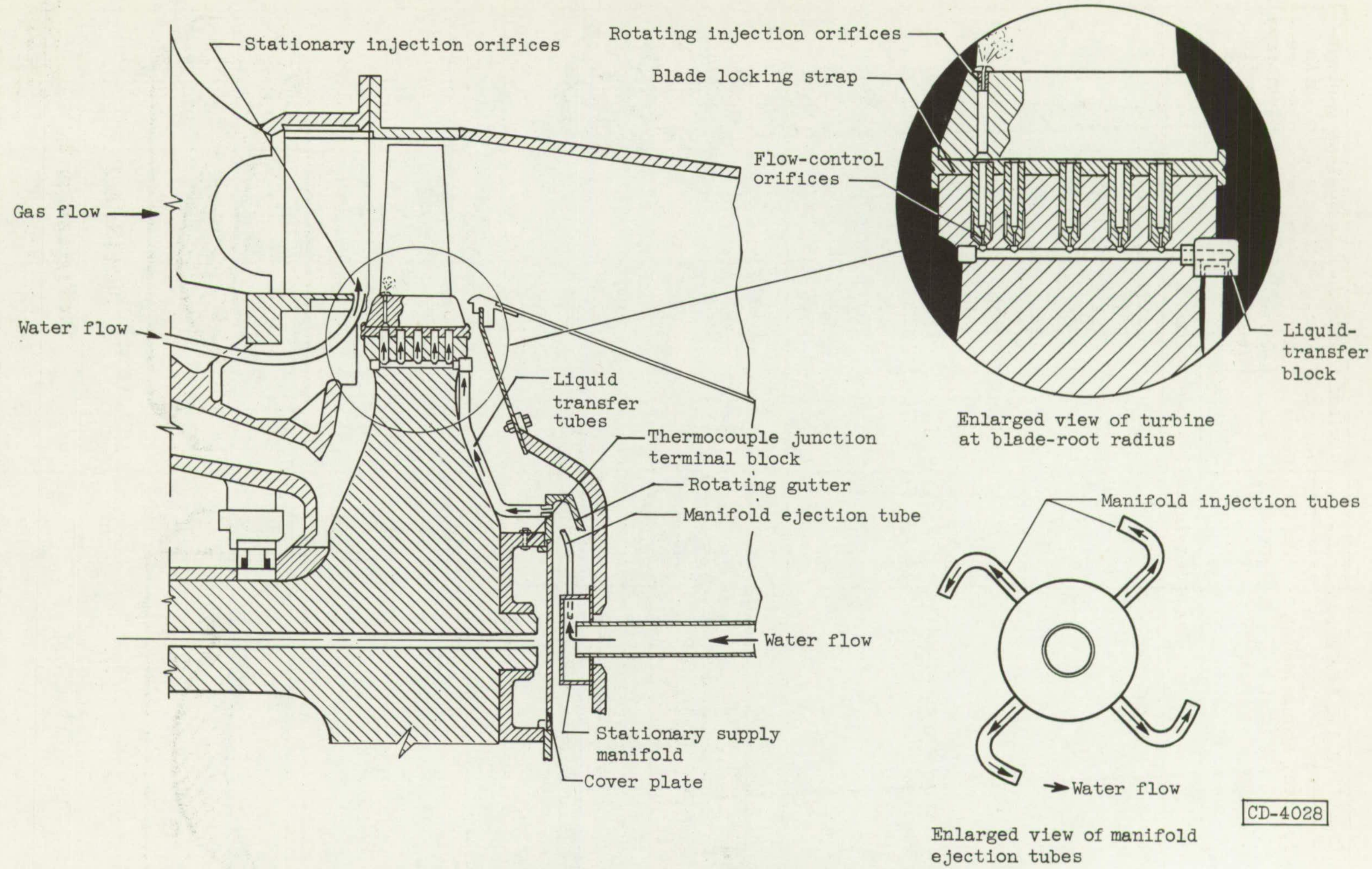



Figure 1. - Sectional view of engine showing methods of water-spray injection from orifices located in the turbine rotor-blade bases and from stationary orifices in stator diaphragm.

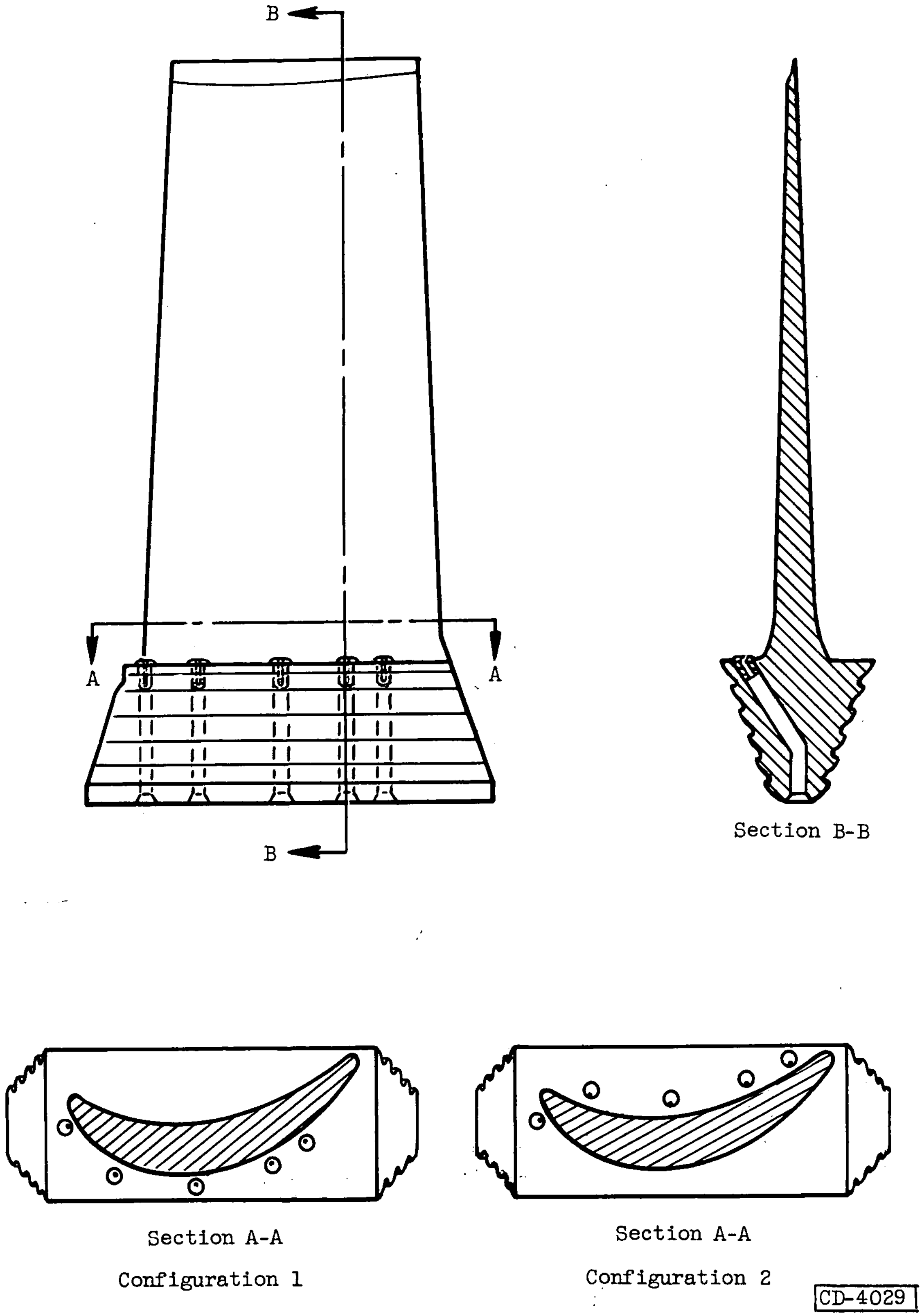



Figure 2. - Types of rotating injection configuration investigated.

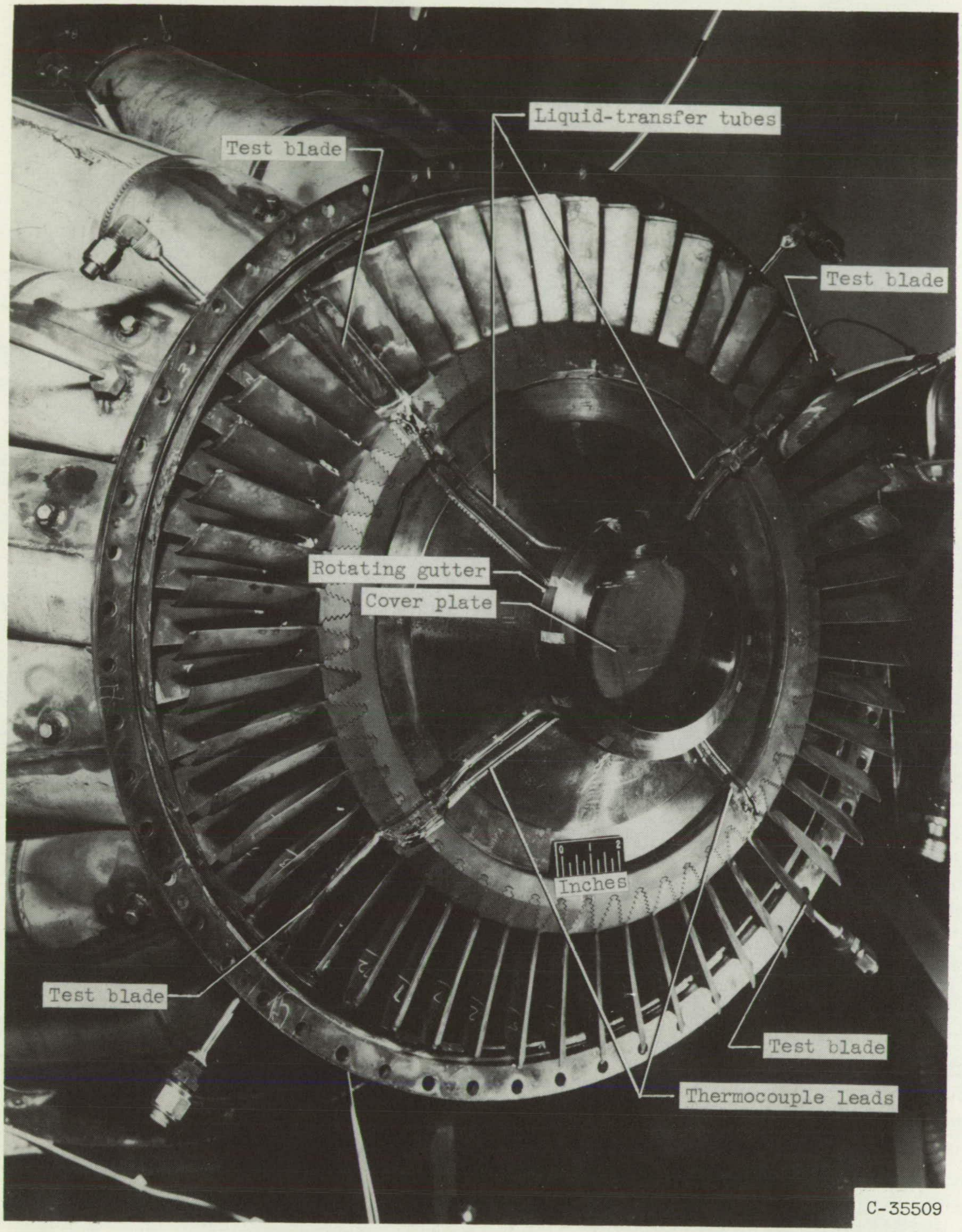



Figure 3. - Engine installation illustrating portions of rotating injection system.

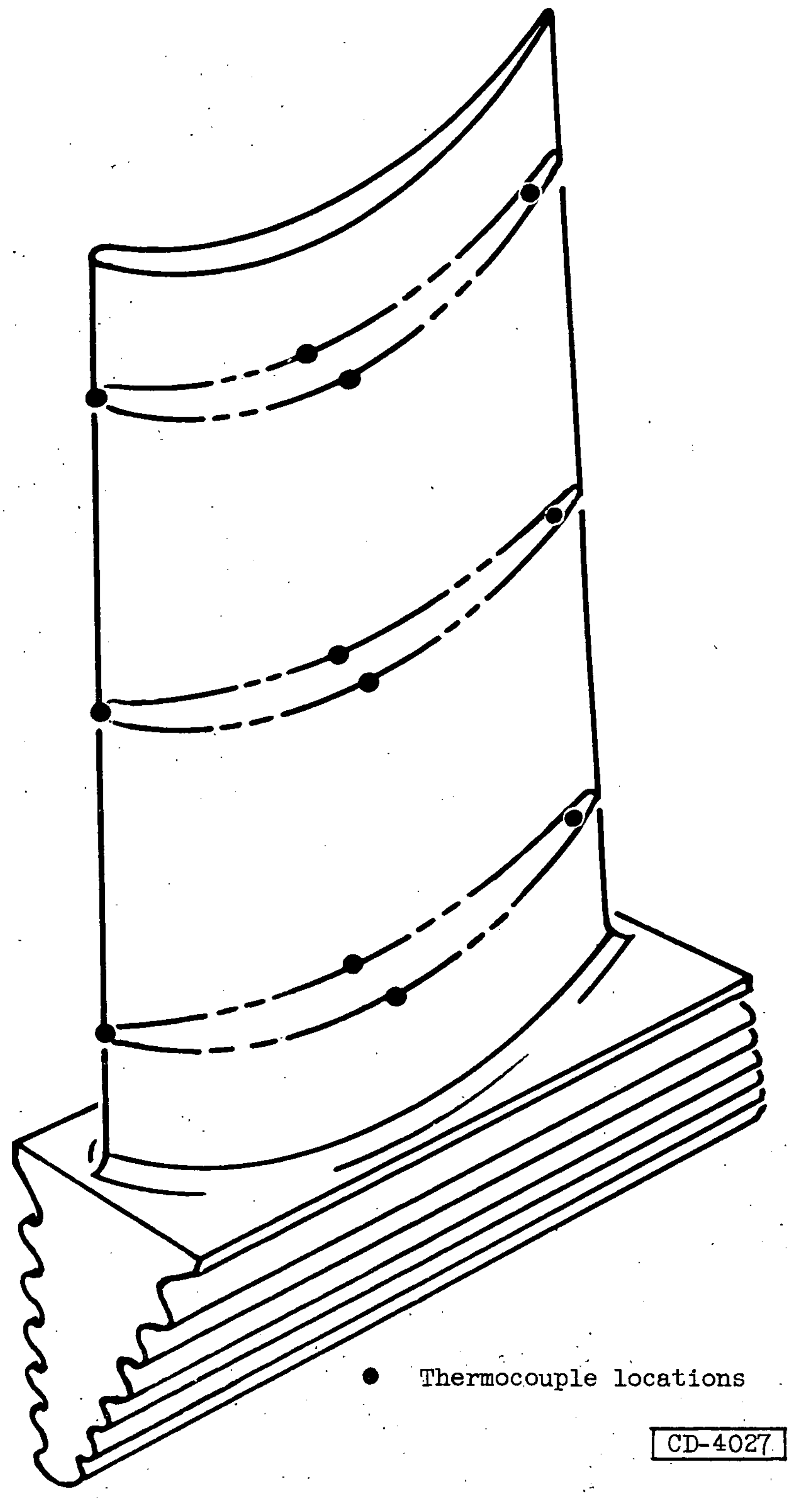



Figure 4. - Thermocouple positions on turbine rotor blades.

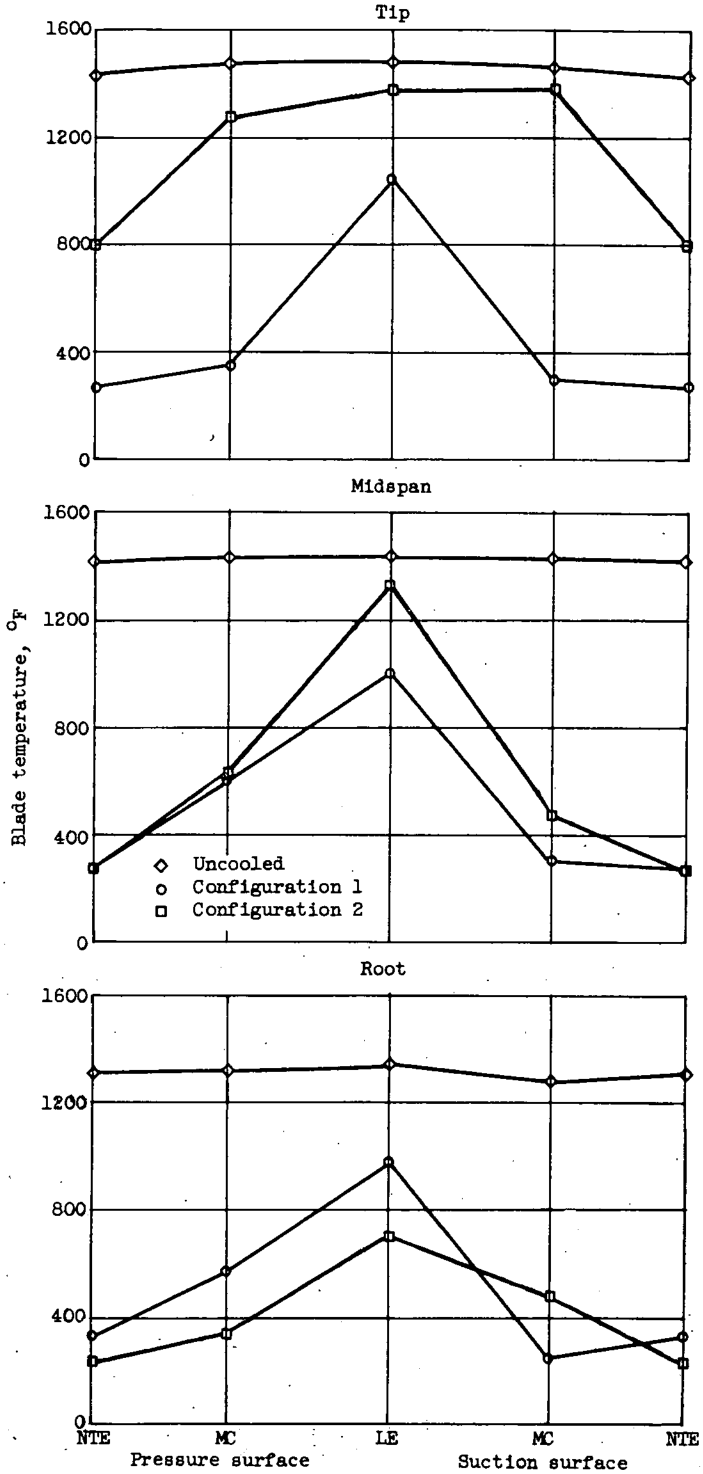



Figure 5. - Comparison of blade chordwise temperature distributions obtained at rated engine speed and equivalent coolant-to-gas flow ratio of approximately 0.035 with the two rotating injection configurations investigated. Near trailing edge, NTE; midchord, MC; leading edge, LE.

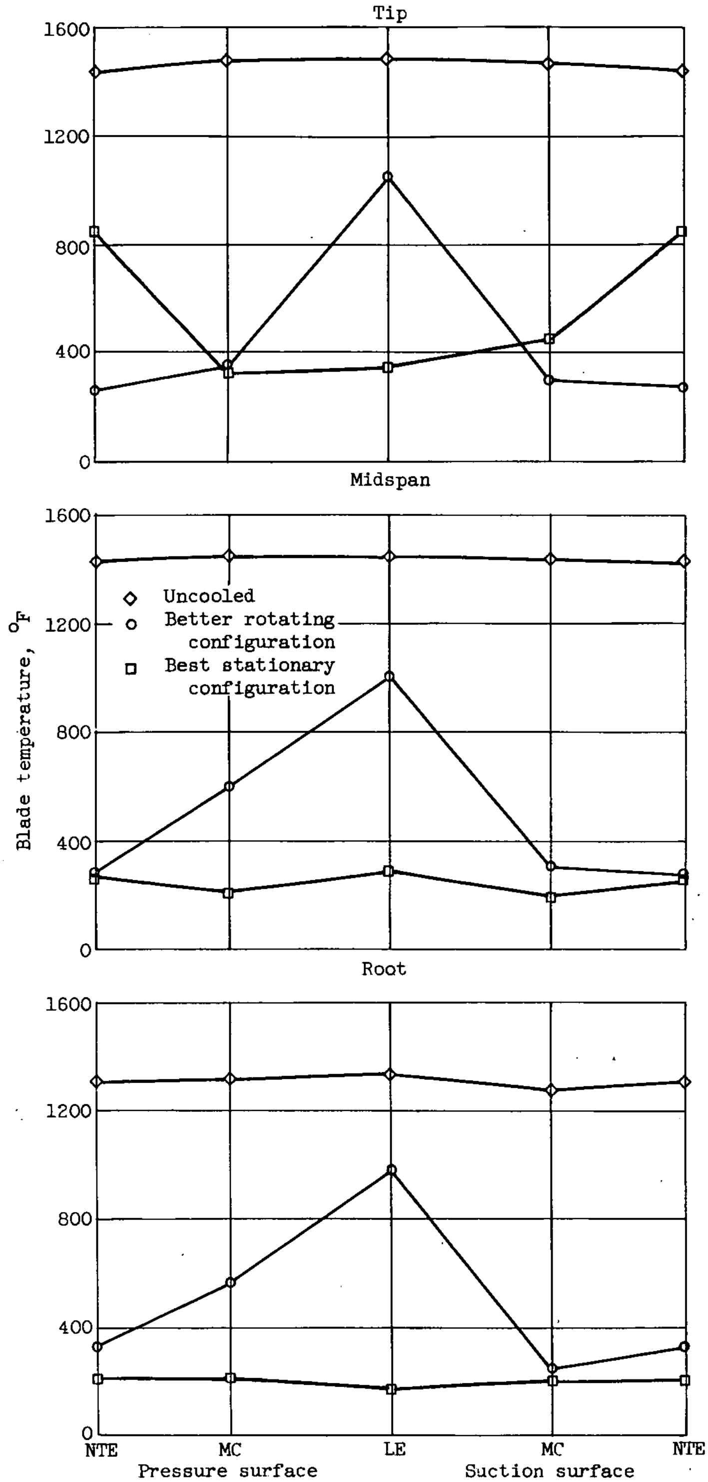



Figure 6. - Comparison of blade chordwise temperature distributions obtained at rated engine speed and equivalent coolant-to-gas flow ratio of approximately 0.035 with more favorable rotating injection configuration and most favorable stationary injection configuration. Near trailing edge, NTE; midchord, MC; leading edge, LE.

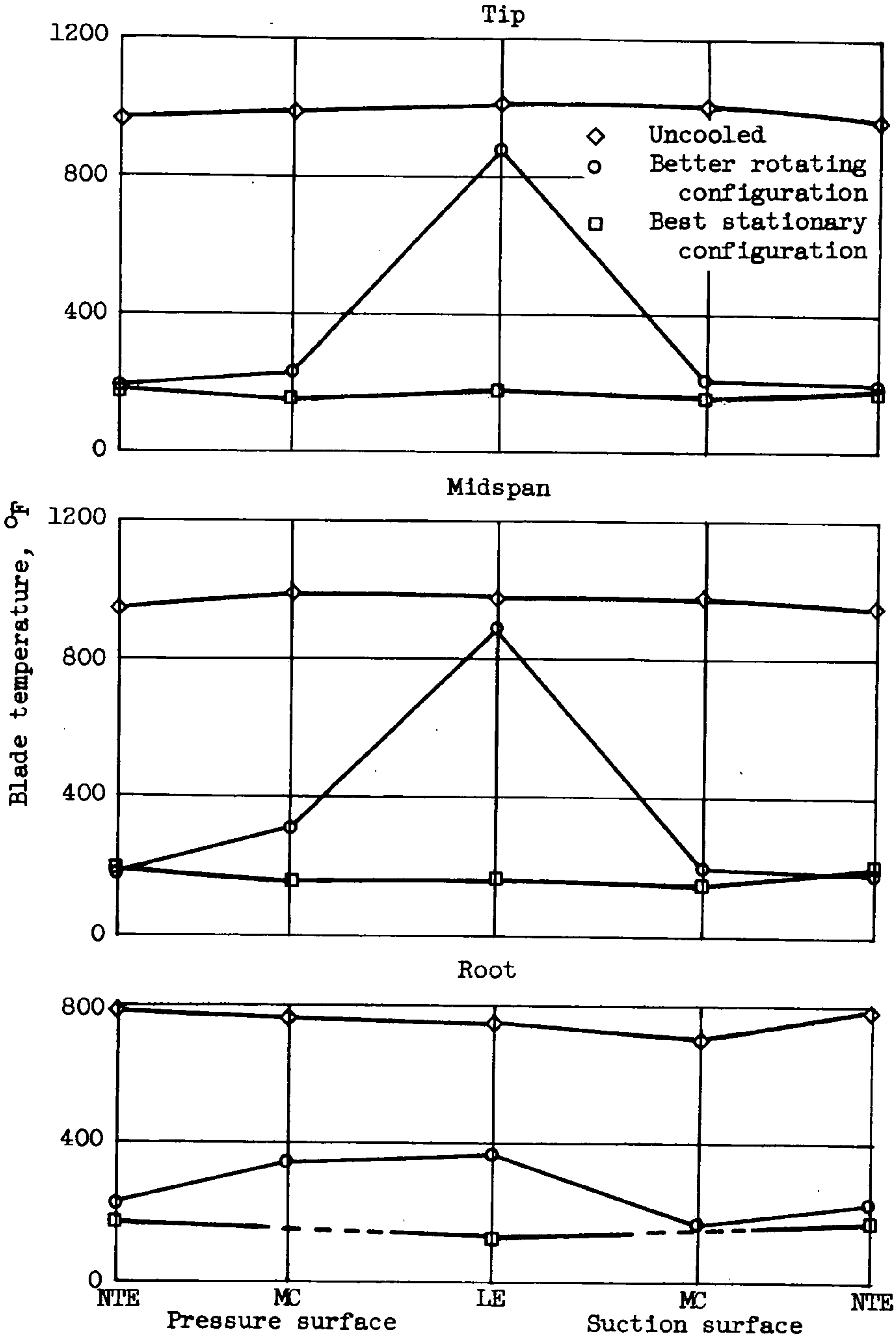
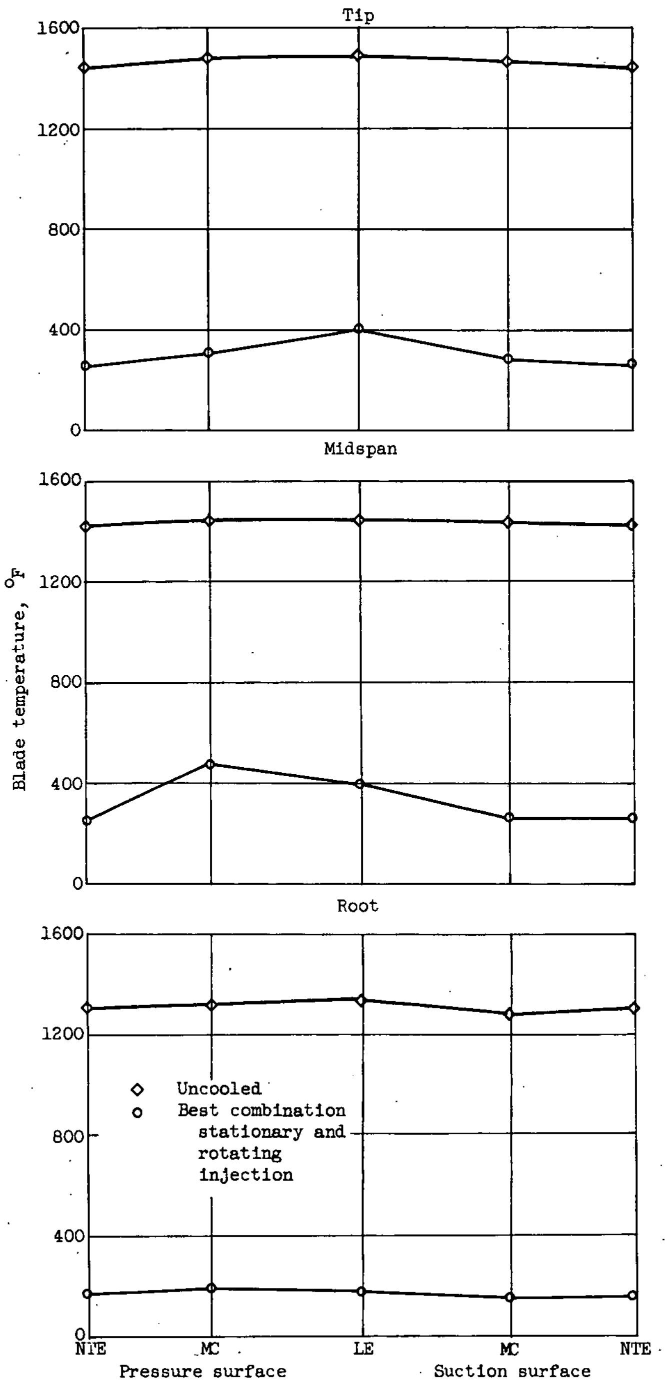
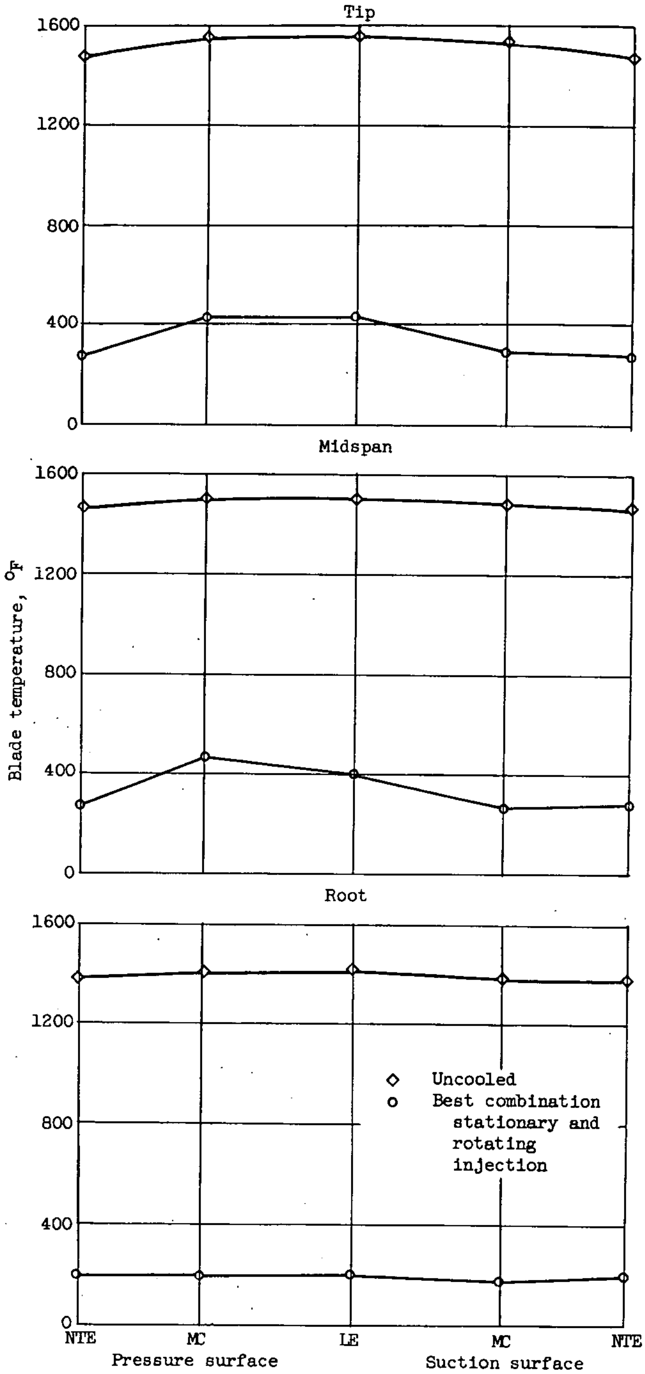



Figure 7. - Comparison of blade temperature distributions obtained with rotating and stationary configurations at equivalent coolant-to-gas flow ratio of approximately 0.035 and engine speed below rated (8000 rpm). Near trailing edge, NTE; midchord, MC; leading edge, LE.




(a) Turbine-inlet gas temperature, 1600° F.

Figure 8. - Blade chordwise temperature distributions obtained at rated engine speed and a total equivalent coolant-to-gas flow ratio of approximately 0.035 with an injection configuration consisting of a combination of rotating and stationary injection orifices. Ratio of rotating equivalent flow to stationary flow, approximately 2:1. Near trailing edge, NTE; midchord, MC; leading edge, LE.




(b) Turbine-inlet gas temperature, 1700° F.

Figure 8. - Concluded. Blade chordwise temperature distributions obtained at rated engine speed and a total equivalent coolant-to-gas flow ratio of approximately 0.035 with an injection configuration consisting of a combination of rotating and stationary injection orifices. Ratio of rotating equivalent flow to stationary flow, approximately 2:1. Near trailing edge, NTE; midchord, MC; leading edge, LE.

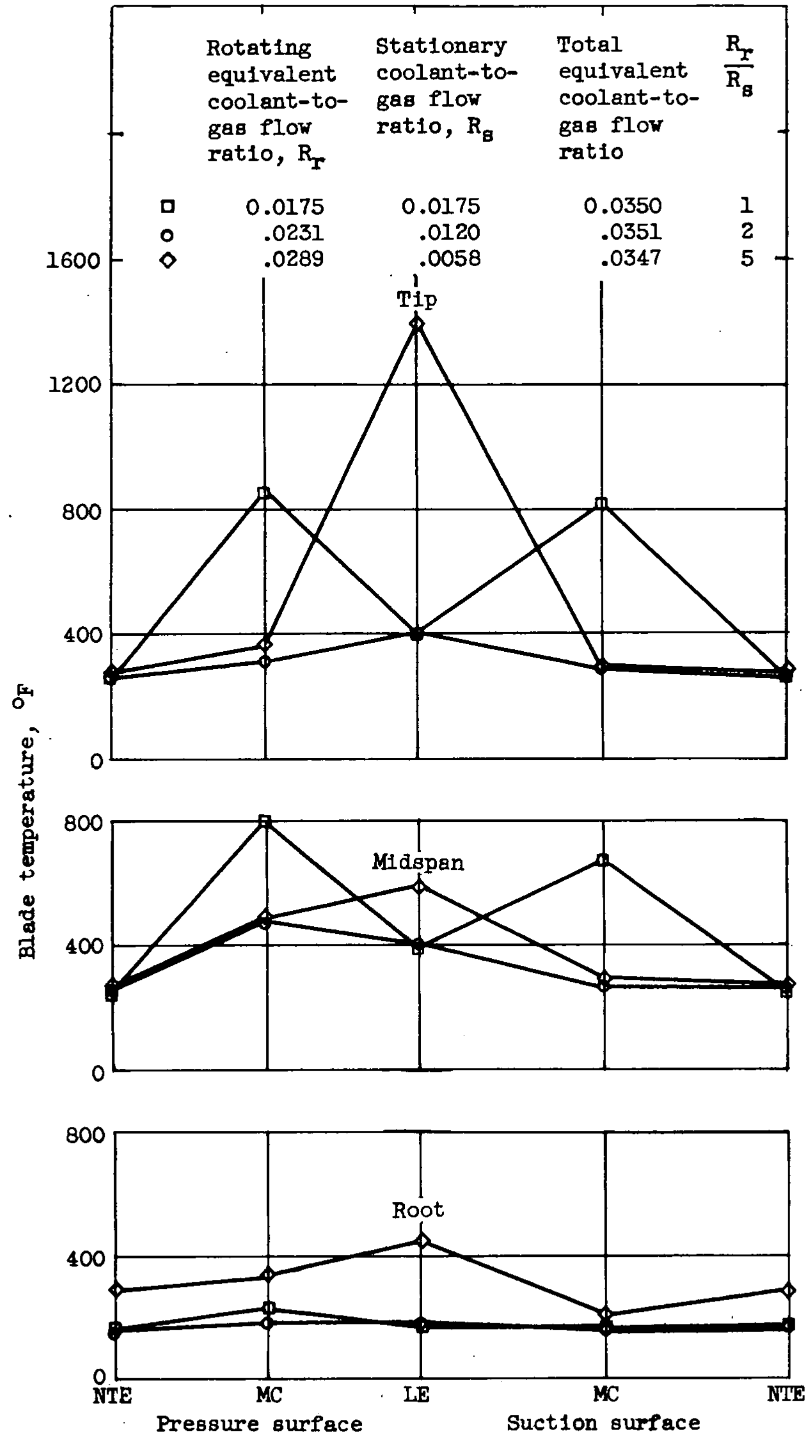



Figure 9. - Blade temperature distributions obtained with changes in coolant-to-gas flow ratio through stationary and rotating portions of most satisfactory combined rotating and stationary injection configuration while maintaining a total equivalent coolant-to-gas flow ratio of 0.035. Near trailing edge, NTE; midchord, MC; leading edge, LE.

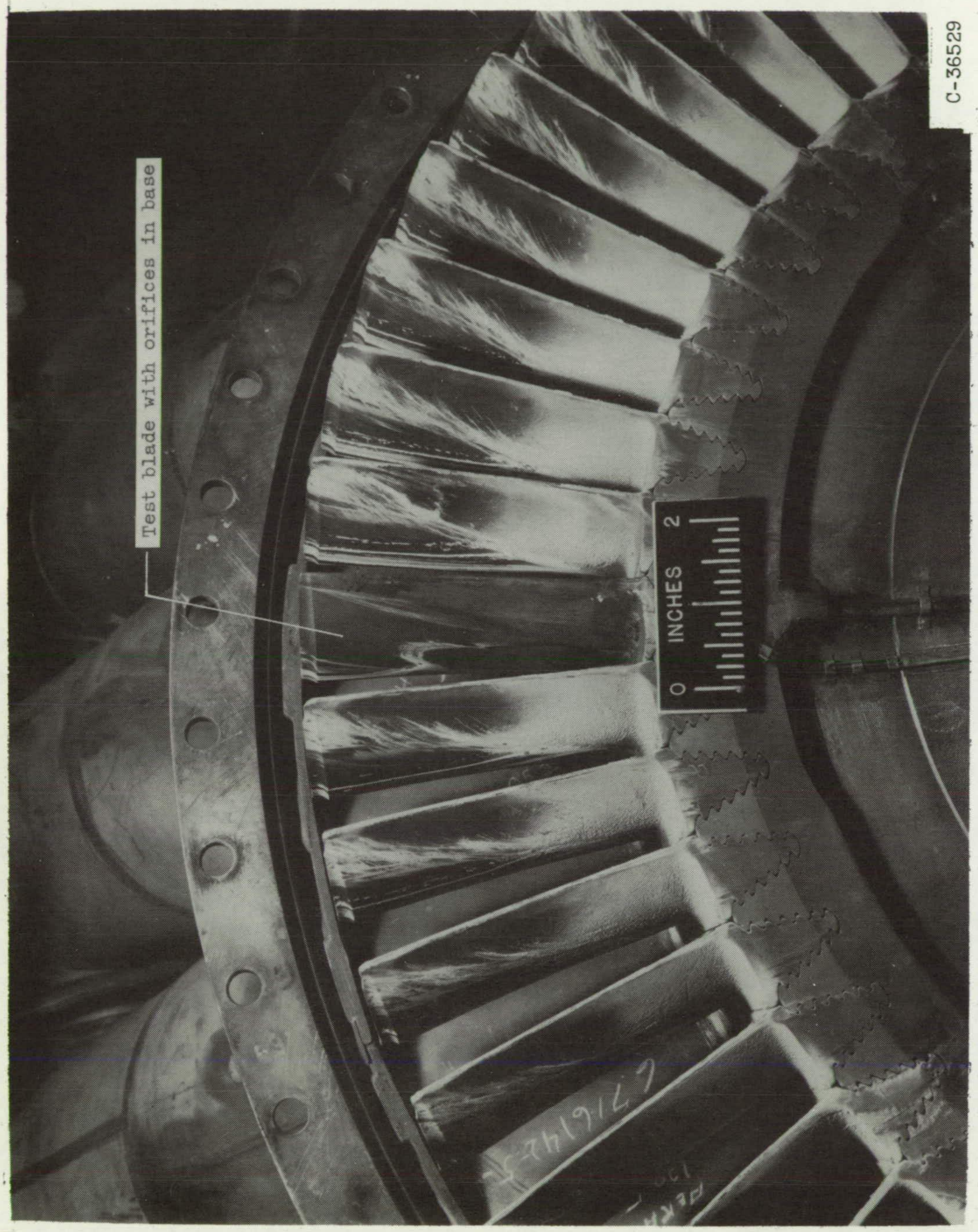



Figure 10. - Turbine blades after rated-speed spray-cooled engine operation with a configuration consisting of a combination of rotating and stationary orifices.